

DESICCANT DEHUMIDIFICATION TECHNOLOGY

Thomas E. Durbin

CECER-UL-U

(217) 398-5543

Michael A. Caponegro, P.E. (Retired)

CECER-UL-U

EXECUTIVE SUMMARY

Desiccants have been used in the manufacturing industry for over 50 years, but have only recently entered the Heating, Ventilation, and Air Conditioning (HVAC) arena. While the private sector has shown an increasing interest, very few military installations have installed desiccant systems, and only then in specialized applications. Depending on climate and facility loading, a high percentage of a building's cooling load can be latent (moisture) load. This portion of the cooling process requires conventional cooling equipment to operate at temperatures low enough to cool the air to its dew point temperature, where dehumidification via condensation on the coils begins. It may then be necessary to reheat the air to a comfortable temperature before it enters the occupied space.

A desiccant dehumidification system (DDS) uses a desiccant to remove water from air. Due to the high temperature at which a desiccant wheel operates, the unit then cools the air with a heat exchange mechanism such as a thermal wheel or heat pipe. Removing moisture from air decreases the amount of energy needed to cool the air supplied to the user and increases the comfort level in the conditioned space. Using desiccant systems greatly reduces that moisture accumulation in ducts and around coils, inhibiting the growth of molds and the formation of mildew.

While research in desiccant dehumidification technology development has continued for several years, commercial applications of desiccant dehumidification technology have been limited in the past by material and manufacturing considerations. Presently, desiccant dehumidification systems capacities up to 30,000 cfm are nearing commercialization. Since these systems are heat driven and not electrically driven, they can reduce site peak electrical demand and levelize utility loads, allowing more efficient power plant operation. Reduced chiller loads, reduced electricity peak demand, and elimination of air reheating re-

quirements combine to reduce energy costs. Desiccant dehumidification systems can reduce or eliminate the use of harmful CFCs in the HVAC system by conditioning air with natural gas or liquid propane gas (LPG) as the primary fuel.

Desiccant dehumidification systems may offer advantages for military applications over other energy supply options by providing increased force readiness, greater reliability, humidity control for areas with sensitive material and equipment, reduced environmental impact, and energy cost savings.

PRE-ACQUISITION

TECHNOLOGY DESCRIPTION AND APPLICATION

The "Conventional" Air Conditioning/Ventilation Process

Conventional air conditioning systems are typically controlled by a thermostat (or some similar receiver/controller combination). Controls are set to keep the space dry bulb temperature from exceeding the thermostat setpoint. To maintain the setpoint, conditioned air is typically introduced into the space approximately 20 °F lower than the setpoint, so that the conditioned air can absorb the "sensible" heat entering the space. Having "absorbed" this heat, air from the space is drawn back to the air handling unit, where its temperature is again decreased before being supplied back to the space. The temperature decrease is accomplished by the returned air being drawn through (or blown through) a cooling coil within the air handling unit. The coil is typically a specially designed finned-tube heat exchanger that contains a relatively cold circulating fluid (usually chilled water or a refrigerant) into which heat from the air is transferred. This situation is often more complicated by the fact that some outside air is then mixed with the returned air from the space, and that mixture is cooled by the coil. The most common reason for introducing outside air is to provide ventilation for the occupants of the space.

As the cooling coil reduces the dry bulb temperature of the air so that the air, in turn, will provide sensible cooling for the space, the dry bulb temperature of the air is reduced almost to its dew point temperature. In fact, a considerable portion of the air actually reaches saturation due to its contact with, or proximity to, the cooling coil, which has a temperature considerably lower than the air's dew point temperature. As a result, water condenses from the air on to the coil. Judicious selection of airflow velocities (< 500 ft/minute) will allow the condensate to drip into a collection pan from which it will drain instead of being blown through the ductwork.

The described process, which began with the objective of keeping the dry bulb temperature of a space from exceeding a thermostatic setpoint, produces a condition where the air introduced is not only cooler, but also drier. One device, the cooling coil, has performed dual service by both lowering the dry bulb temperature of the air and reducing its moisture content. The moisture removal has not been incidental or accidental; the cooling coil was selected based on its capability to remove the space and outside air sensible and latent (moisture) loads estimated to occur on a "design day."

Potential Problems with the "Conventional" Process

"Design day" conditions are generally defined as the dry bulb temperature and its mean coincident wet bulb temperature that are equaled or exceeded 2.5 percent of the time, on the average, during June, July, August and September (months applicable for DOD installations in the contiguous United States). Generally, under design day conditions, the conventional process described previously can produce satisfactory conditions of dry bulb temperature and relative humidity within the space (design and selection of the air conditioning system components were based on design day conditions). For an appreciable amount of time, off-design conditions prevail during which the proportion of the latent load to the total outside air cooling load is likely to increase, compared to the ratio at the design day conditions. Table 1 lists outside conditions for a DOD site:

Table 2 lists latent cooling ratio for the outside air conditions, assuming, for simplicity, unity flow for the above conditions, the sensible, latent and total loads. The numbers in Table 2 should not be construed to mean that the conventional process will necessarily provide poor indoor environmental conditions at off-design conditions. Space loads may predominate over outside air loads and the sensible heat ratio for the coil may stay relatively constant over the range of outdoor air conditions. The numbers do suggest there could be a problem, particularly for facilities where the outdoor air load on the coil is a large part of the coil total cooling load. It has become more likely for this to happen following the issuance of ASHRAE Standard 62-1989, which calls for more outdoor air than previously required (as much as 20 cfm/person) for ventilation.

Trying to improve indoor air quality retroactively through compliance with the ASHRAE standard can be futile in many cases because the existing equipment lacks the capacity to handle the

TABLE 1
OUTDOOR CONDITIONS

DRY BULB TEMP (BIN AVERAGE (°F))	WET BULB TEMP (°F)	SPECIFIC HUMIDITY ANNUAL	
		(GRAINS/LB AIR)	HOURS
102	74	81.1	4
97	74	89.2	49
94	75	100.1	DESIGN DAY
92	74	97.3	250
87	72	93.8	479
82	71	96.3	659
77	69	93.5	921

TABLE 2
LATENT COOLING RATIO FOR OUTSIDE AIR CONDITIONS

SENSIBLE LOAD	LATENT LOAD	TOTAL LOAD	LATENT/ TOTAL
1.08 X (102 - 75) = 29.16	0.68 X (81.1-65)=10.95	40.11	0.273
1.08 X (97 - 75) = 23.76	0.68 X (89.2-65)=16.46	40.22	0.409
1.08 X (94 - 75) = 20.52	0.68 X (100.1-65)=23.87	44.39	0.538
1.08 X (92 - 75) = 18.36	0.68 X (97.3-65)=21.96	40.32	0.545
1.08 X (87 - 75) = 12.96	0.68 X (93.8-65)=19.58	32.54	0.602
1.08 X (82 - 75) = 7.56	0.68 X (96.3-65)=21.28	28.84	0.738
1.08 X (77 - 75) = 2.16	0.68 X (93.5-65)=19.38	21.54	0.900

additional load imposed by the increased amount of (humid) outside air. Further, the sensible heat ratio for the coil will likely differ, perhaps significantly, even for design day conditions, since the outdoor air load will be a larger proportion of the total cooling load. The Air Force (and ASHRAE) have recognized that, for numerous locations, operational problems at off-design conditions may likely occur using the "design day" concept explained above as the basis for air conditioning design. In an attempt to minimize or eliminate, these problems, the Air Force is (by contract) restructuring the data contained in the document Engineering Weather Data (AFM 88-29, TM 5-785, NAVFAC P-89) to, among other things, highlight for designers those locations where chronically high outdoor humidity levels need to be addressed during the design process.

Note that the conventional process can be modified, with some increase in control complexity and first cost to achieve improved

indoor environmental conditions under off-design outdoor conditions. The modification essentially overcools the air in response to a call for dehumidification from a humidistat (or by turning down a thermostat), then reheating the cold dry air as necessary to ensure that the thermostat dry bulb temperature setpoint is not exceeded. As noted, this scheme increases the controls complexity and first cost. The primary increase in cost, however, results from the cooling system running longer to dehumidify the air and the air subsequently requiring reheat. This type of modification is seldom employed due to the additional costs just cited. It is used for spaces where precise humidity control is essential, such as laboratories, clean rooms, and hospital operating rooms. It would be extraordinary (and expensive) for reheat to be employed for an office building (or for numerous other types of facilities). For those types of facilities, off-design outdoor conditions may well result in a somewhat humid indoor environment. Alternatively, to address occupant complaints of discomfort, the thermostat setpoint may be lowered, reducing the indoor humidity level. without reheat control, this action can lead to complaints because the space will feel too cold. Poor indoor environmental conditions often result in worker/occupant discomfort and decreased productivity.

Another potential problem with the conventional process is that of microbial and fungal growth in condensate drain pans. These can be carried into the ductwork and deposited where further growth can occur. Microbes and bacteria can be introduced into the space from breeding grounds in the pan or ductwork, causing occupant discomfort and possible allergic reactions or illness. Reheat will not solve this potential problem. Biological fouling of ducts may pose a serious problem in sensitive spaces such as operating rooms requiring a sterile environment. To summarize, potential problems with the conventional process are:

- difficulty in providing satisfactory indoor environmental conditions when off-design outdoor conditions are experienced
- first cost and operating expense increase when the conventional system is modified with reheat control to provide satisfactory environmental conditions when off-design outdoor conditions
- difficulty in modifying existing conventional systems to handle additional outdoor air cooling load resulting from increased ventilation rates called for by ASHRAE Standard 62-1989
- indoor air quality problems due to microbial or fungal growth in condensate drain pans and ductwork.

Desiccant Dehumidification Offers Possible Solutions

Desiccant dehumidification equipment can, in many cases, address the problems cited above for the conventional process. There are basically two types of desiccants (materials that can directly remove moisture from the air):

- a solid material such as silica gel or molecular sieve that is deposited on the flutes of a rotating honeycomb wheel
- a hygroscopic liquid that is sprayed into the air stream to remove moisture.

The dehumidification process is similar for each type. For simplicity, the following discussion is focused on solid desiccant equipment. Figure 1 shows the desiccant wheel operation. Humid process air (which will be supplied to the occupied space) passes through the desiccating portion of the desiccant unit where the air is dehumidified. The process air experiences a significant increase in its dry bulb temperature due to: (1) the latent heat released on condensation of the removed water, and (2) the temperature of the wheel due to the heat required to regenerate the desiccant. The desiccant wheel, belt- or chain-driven by an electric motor and laden with moisture from the process air, rotates slowly (~ 0.2 revolutions/ minute) into a separate hot air stream, which will remove that moisture so that the "regenerated" desiccant can absorb moisture when it rotates back into the humid process air stream.

Figure 2 shows the desiccant wheel relative to the other components typically provided to make the system work. Note that two modes of operation are shown: RECIRCULATION and VENTILATION. The choice as to which mode is preferable depends on first cost differences, the specific building application, utility rates, and climate. Regardless of the mode of operation, separate fans, one to move the process air and the other to move the regenera-

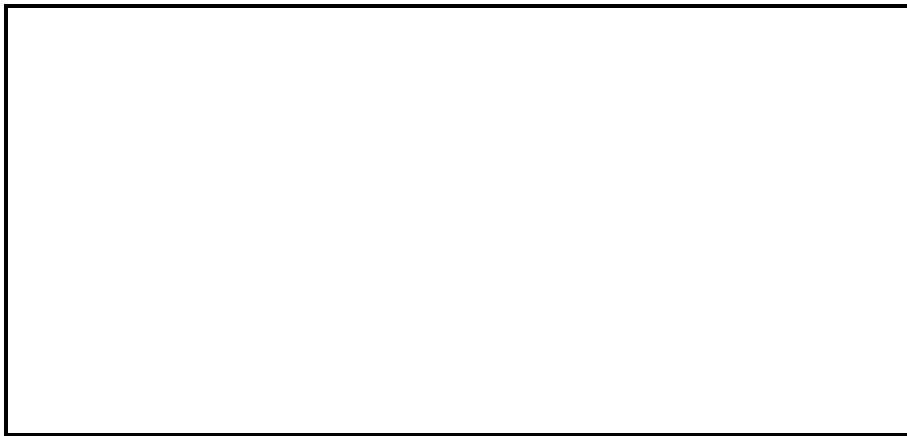


FIGURE 1. DESICCANT WHEEL OPERATION.

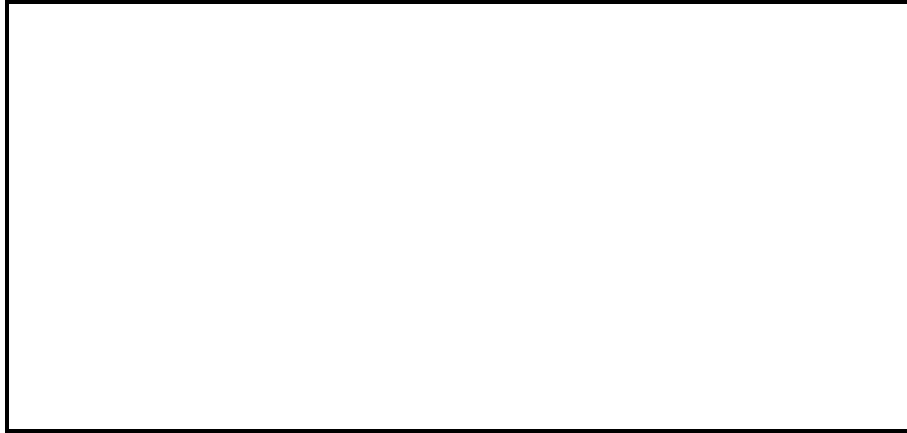


FIGURE 2. DESICCANT WHEEL RELATIVE TO OTHER COMPONENTS.

tion air, are used. On the process air side, the humid process air typically enters the desiccant at state 1 and emerges at state 2, dryer and hotter. The hot, dry process air at state 2 then passes through a heat exchanger where it is sensibly cooled to state 3. Usually, the process air at state 3 is too warm to deliver to the space and achieve sensible cooling. Consequently, some final element such as a direct evaporative cooler or cooling coil is used to condition the air to state 4 before its entry into the space.

On the regeneration air side, exhaust or outside air at state 5 passes through a direct or indirect evaporative cooler to reach the condition at state 6. This air is cooled so that it can, in turn, cool the heat exchanger, after which the air is at state 7. The air at state 7 is then heated by the regenerator to the much higher temperature at state 8. From state 8 to state 9, the air regenerates the desiccant. It is not readily obvious why air the used to regenerate the desiccant should be initially cooled. One would think substantial energy waste might result. However, the process uses relatively inexpensive evaporative cooling. This cooling allows the heat exchanger to cool the process air more effectively. The heat from the process is transferred to the regeneration air, increasing its temperature and reducing the amount of energy that must be supplied by the regenerating heater. The heat exchanger may be a plate-type heat exchanger, thermal wheel, or heat pipe, depending on the desiccant unit manufacturer. (The latter two types are the most common.) For all types, the energy transferred is principally sensible heat. The thermal wheel is driven in a manner similar to the desiccant wheel, but rotates faster (10 to 20 revolutions/minute). Some potential process air mixtures that may be employed are:

- 100% outside air, all desiccated
- only outside air desiccated, then mixed with return air
- outside air and return air mixed, with the mixture desiccated.

In most cases, regardless of the source of the air to be desiccated, some final dry bulb temperature reduction will be required, usually requiring a cooling coil. However, this coil will likely not have to do any further dehumidification. Basically, using the desiccant for dehumidification has enabled the decoupling of the dry bulb cooling and dehumidification processes, allowing the cooling coil to do sensible cooling with minimal (if any) latent cooling. This decoupling enables the desiccant system to address the problems cited earlier for the conventional system, as described in the following paragraphs. The desiccant system can provide the dehumidification required to meet the space's latent load for the process air under all outdoor air conditions. The cooling coil will provide the required sensible cooling and remove any residual moisture so the air introduced into the space will also meet the space's sensible cooling load.

The desiccant unit itself is generally large and heavy, and will, if anything, result in increased first cost compared to adding reheat to a conventional system. However, installing a desiccant can result in reduced operating cost compared to a conventional system with reheat. This is more likely where the cost of electricity is high compared to natural gas (fuel used as the energy source for desiccant regeneration). The user needs to bear in mind the fact that electrical billing for DOD facilities typically has two components, an energy charge and a demand charge. The demand charge is usually a significant portion of the total cost for electricity. When an electrically powered water chiller or electrically powered direct expansion equipment would otherwise be used to provide latent cooling, a desiccant used for that purpose will reduce both electrical demand and electrical energy consumption, and the associated cost for each. Energy consumption for reheat would be eliminated. Potential reduction in evaporator temperature to ensure adequate moisture removal would not be necessary. A dry cooling coil to enhance heat transfer may actually permit an increase in evaporator temperature without sacrificing sensible cooling capability. With the air in the space drier due to the desiccant's deep dehumidification capacity, it may also be possible to increase the dry bulb temperature setpoint for the space without sacrificing occupant comfort.

Latent cooling using desiccation may be almost free in circumstances where waste heat, such as that from a natural gas engine-driven chiller, may be used for desiccant regeneration. Ancillary environmental benefits can result when latent cooling is accomplished through desiccation, instead of by subcooling the air stream using electrically-powered equipment. This will occur when the primary energy source for desiccation is clean-burning natural gas, and the electrical energy that would otherwise be required for the electrically-powered equipment is from a coal- or fuel oil-fired power plant.

Installing a desiccant unit may well be the least-expensive way to retrofit a facility to ensure compliance with ASHRAE Standard 62-1989. Increasing the amount of ventilation air will generally increase the sensible and latent cooling loads imposed on the cooling coil. The exception, of course, would be when outside air conditions and a facility cooling load warrant air-side economizer operation. The latent cooling capacity of the desiccant can "free-up" equivalent capacity in the chiller or direct-expansion equipment, allowing that equipment to possibly meet the additional sensible cooling loads arising from the increased ventilation air flow. Similarly, the cooling coil may well experience no increase in total load, with the increase in sensible load from the increased amount of outside air negated by the removal of most, if not all, of the outside air latent load it formerly had to remove plus the additional latent load due to the increased amount of ventilation air. Further, the cooling coil should perform more effectively since sensible heat ratios will invariably be high.

The foregoing discussion leads to the conclusion that microbial or fungal growth in the condensate drain pan and ductwork should be eliminated since the cooling coil will be a virtually dry coil for the vast majority of the time.

Types of facilities where desiccant technology may well be applied to performance and economic advantage include refrigerated warehouses, ice rinks, supermarkets, laboratories requiring close tolerance on relative humidity and/or with significant makeup air requirements, educational facilities, humidity-controlled warehouses, lodging facilities, commissaries, and medical facilities (particularly operating rooms).

COSTS AND BENEFITS

The main factors that will determine the amount of energy and energy cost savings achievable by installing a desiccant system have been implicitly covered already. The desiccant unit will require electrical energy for the process and regeneration fan

motors, the fractional horsepower motors required to drive the desiccant wheel and (if used) thermal wheel heat exchanger, the hot water circulating pump motor when hot water is used for desiccant regeneration, and for any evaporative cooler water pump motors. The largest energy use by the desiccant unit is for the heat required to regenerate the desiccant material. Generally, this heat is produced by combustion of natural gas. To undertake an accurate analysis, the user will have to make a preliminary selection of a desiccant unit suitable for the application and obtain manufacturer's data regarding motor horsepower and regeneration energy requirements for the anticipated modes of operation. Another cost that must be included is the cost to provide the final sensible cooling that may be required to decrease the dry bulb temperature of the desiccated process air stream prior to its being introduced into the space. The user must ensure that the cost for electrical demand is included. The demand charge is a cost for electrical power (kW), not electrical energy (kWh). The desiccant unit's thermal energy and electricity costs would be weighed against the energy and demand costs for the conventional system to deliver the same amount of air to the space at the same conditions. To ensure a fair comparison, costs should be included for any subcooling and reheating that would be required for a modified conventional system to provide the same indoor conditions as the desiccant system would provide, for all outdoor conditions occurring when dehumidification and/or sensible cooling would be required to provide those indoor conditions.

EXAMPLE COST SUMMARY

This example is for a desiccant unit placed on an Avionics facility in Jacksonville, FL. The local natural gas cost is \$0.35/therm and the local electricity cost is \$0.068/kWh. The electrical demand charge is part of the base rate (\$0.068/kWh), so no separate cost for demand is in the cost summary. The desiccant unit capacity is 5670 cubic feet per minute (cfm) and that amount of desiccated air is mixed with 15,130 cfm of return air. This system operates approximately 7050 hours per year. The desired conditions in the conditioned space are 75 °F and 42 percent relative humidity (RH). The return air is typically 78 °F and 62% RH. The energy use and cost comparison is between a conventional cool/reheat system that uses steam for reheat at a cost of \$14.75/MBtu and a cooling system retrofitted with a desiccant dehumidification unit to dehumidify the outside air. The desiccant unit energy consumption is based on data from Engelhard/ICC. The desiccant unit is expected to last 20 years, with a major overhaul scheduled for the tenth year for life cycle cost calculations. The cost of the 5670 cfm unit is approximately \$61,000. Installation costs are estimated to be \$75,000 for a roof-mounted unit of this size. The maintenance require-

ments are estimated at 100 hours per year for this unit. Maintenance labor costs, at a cost of \$35.00/hour, would be \$3500/year.

Table 3 was developed using a preliminary energy and economics analysis spreadsheet created for use in screening of candidate sites for desiccant technology application. Evaluation of potential projects with this screening tool can be performed by USACERL. The primary inputs necessary for this screening include building function, size of area, local utility rates, local weather data, description of current system, and conditioned space requirements.

TABLE 3
COST SUMMARY OF CONVENTIONAL VS. DESICCANT SYSTEM

PARAMETER	CONVENTIONAL SYSTEM	WITH 5670 CFM DESICCANT
ELECTRICITY RATE (\$/KWH)	0.068	0.068
NATURAL GAS RATE (\$/THERM)	0.35	0.35
ANNUAL ELECTRICITY (KWH)	674,327	544,911
ANNUAL NATURAL GAS (MCF)	0	2,000
ANNUAL ELECTRICITY COST (\$)	45,517	36,781
ANNUAL NATURAL GAS COST (\$)	0	7,080
ANNUAL REHEAT COST (\$)	23,933	0
TOTAL ANNUAL COST (\$)	69,450	43,861
ANNUAL SAVINGS (\$)		25,589

The payback period on the investment is then:

[Initial Cost, Installed]/[Annual Energy Savings - Annual Labor Cost]

[\$61,000 + \$75,000]/[\$25,589 - \$3500] = 6.16 years.

UTILITY AND SPACE REQUIREMENTS

In planning possible use of desiccant dehumidification equipment, the user must consider whether electricity, water (for evaporative cooling) and an energy source for desiccant regeneration (usually natural gas) will be available at the site in sufficient quantity and (for natural gas) at the proper pressure. If not already available as required, the user must consider whether utilities can be brought to the site in the quantities and at the pressures required, and how much it will cost. Other siting considerations include unit size and weight, and clearances required for safety, maintenance, and adequate air flow. The latter information is usually available from reputable vendors. Of course, before considering the siting issues, the user must examine performance data supplied by various desiccant vendors and at least tentatively select models that will provide the

degree of dehumidification required for the application under consideration. Desiccant units can be roof-mounted (with appropriate curbs supplied by the vendor) or ground-mounted. If roof-mounted, provisions should be made for safe access to the roof. The structural strength of the existing roof and supporting framing must be checked for adequacy. Aesthetics are invariably a consideration for either roof- or ground-mounted units. Roof-mounted units may have to be located away from the edges of the roof or behind a parapet wall to minimize the unit's visibility. Ground mounting may require the expense of a screen wall or fence. Table 4 lists possible vendors gleaned from the third edition of the Natural Gas Cooling Guide, published by the American Gas Cooling Center.

TABLE 4
DESICCANT VENDOR LIST

COMPANY	PHONE	FAX
AIRFLOW COMPANY	301-695-6500	
DRYOMATIC GENERAL PRODUCTS GROUP	301-631-0396	
ENGELHARD/ICC	215-625-0700	215-592-8299
KATHABAR SYSTEMS DIVISION	908-356-6000	
SOMERSET TECHNOLOGIES INC.	908-356-0643	
MUNTERS CORPORATION	210-651-5018	
DRYCOOL DIVISION	210-651-9085	
SEASONS 4 INC.	404-489-0716	404-489-2938
SEMCO INCORPORATED	314-443-1481	314-443-6921

ACQUISITION/PROCUREMENT

ACQUISITION/PROCUREMENT STRATEGY

The initial step in the typical DOD acquisition process is design accomplished under a design contract. Construction then follows based on the design plans and specifications that have been incorporated into a construction contract. Within DOD, project specifications are usually an assemblage of generic guide specifications edited to address the specific requirements of a particular project.

Guide specifications for particular items of equipment are generally the result of considerable research and experience with different types of equipment intended to perform a given task or function. They are usually based on technical criteria and guidance that have been developed within the Government and refer to standards that industry has developed for the equipment and/or its components. Over time, the guide specification writer eliminates portions of guide specifications that have allowed equipment to be procured and installed that performed inadequately or failed prematurely. Portions of guide specifications dealing with equipment that has performed well are, of course, retained. At this time, the Corps of Engineers is developing guide specifications and technical guidance for desiccants for DOD facilities in general. However, designers of DOD commissaries have been specifying desiccants for their facilities for years and have developed guide specifications for the systems appropriate for their facilities. There are alternative approaches available to the typical design and construction scenario outlined above. An integrated design/build approach may be taken. A Request for Proposals (RFP) is issued which indicates the functional and performance requirements for a project to prospective offerors. The Government then reviews the proposals and selects the one offering the best value in satisfying the requirements in the RFP. This approach is one possible way to install a satisfactory desiccant system. USACERL can provide a sample scope of work and equipment specifications.

DESIGN CONSIDERATIONS

The designer must revisit the system performance considerations mentioned previously. Decisions about the source of the air for desiccation must be made (100% outside air supplied to the space, outside air subsequently mixed with return air or outside air and return air mixed, then desiccated); source of air for regeneration (outside air, exhaust air or a mixture of the two); medium for regeneration (steam, hot water, or products of combustion [direct or indirect]); and method(s) for process air post-cooling.

The designer should thoroughly examine the existing heating, ventilating, and air-conditioning (HVAC) systems already serving the spaces that will be served by the desiccant unit to determine how the unit should interface with the existing equipment, from a control as well as physical standpoint. The sequence of operation and a control diagram for all fans, pumps, and operators for dampers and valves should be provided on the design drawings. Internal controls to be provided as an integral part of the desiccant unit should be specified as such. Ladder diagrams showing safety interlocks and all on/off controls should be

provided. Proper control design, installation, and documentation are paramount if the desiccant unit and the entire HVAC system are to meet the requirements of the spaces to be served, and do so cost effectively.

The designer should indicate in the specifications that complete operation and maintenance manuals are to be provided for the desiccant unit. Manuals will clearly explain the function of each major component of the desiccant unit -- desiccant wheel, regenerator, etc. and indicate maintenance intervals and procedures for all unit components for which maintenance will be required. Manuals will contain control drawings and schematics as outlined in the preceding paragraph. Specifications should also indicate that the contractor and desiccant unit manufacturer will provide training (clearly specifying the duration and number of trainees) regarding operation of the desiccant unit and the HVAC system of which it is to be a part. Such training can be omitted if maintenance will be performed under a service contract. Strong consideration should be given to entering into an extended warranty agreement. The designer must design for maintainability, ensuring proper clearances around the unit in accordance with the manufacturer's recommendations so as not to compromise safety, access, and performance.

CONSTRUCTION CONSIDERATIONS

It is highly recommended that the project specifications require detailed contractor submittals for the desiccant unit itself and the HVAC/desiccant controls. These submittals and all contractor substitution proposals should require "E [Engineer]- level" review and approval or disapproval. Further, it is recommended that the Government contract with the designer to provide these review services as an extension of his design. The designer should also develop the as-built drawings for the project.

POST ACQUISITION

COMMISSIONING

It is recommended that the entire system be tested under normal as well as extreme operating conditions. Simulation of design-day performance and off-design performance should be performed immediately after installation and before final acceptance is issued. The commissioning process can be performed by the customer or by a third party. Written schedules and logs for recording maintenance should be provided and kept near the unit for convenience. Laminated schematics and preventive maintenance guides should also be provided and kept near the desiccant unit.

It is also recommended that the operators attend a detailed training session on the equipment before the customer issues final acceptance of the system. The training should include on-site instruction and written materials, an explanation of the concept of desiccant dehumidification and its role in modern HVAC systems, description of the system components, analysis of the internal operation, recommended preventive maintenance to be scheduled and performed, troubleshooting tips, and a manufacturer's point-of-contact for warranty issues.

OPERATION AND MAINTENANCE ISSUES

Routine maintenance for optimal system performance includes:

1. Inspection and filter replacement at intervals recommended by equipment manufacturers
2. Lubricate desiccant and heat exchanger wheel bearings twice per year
3. Lubricate fan motor bearings twice per year
4. Check/clean evaporator pads at the beginning and end of cooling/heating seasons
5. Check controls and settings twice per year
6. Clean unit, fans, and coils as required by conditions (at least annually)
7. Repair any broken or defective part whenever reported or found (immediately)
8. Report to Post Engineer any problem when found (immediately)
9. Balance system and optimize performance of units based upon loads twice per year
10. Tune burners at least once per year (when applicable).

PERFORMANCE EVALUATION

The performance of a desiccant unit and the HVAC system it operates within can be evaluated by using Energy Management System (EMS) equipment or a separate data logging computer and sensors. Feedback from occupants, measurements of temperature and humidity in the occupied space, and inspection of materials in the occupied space also serve as important indicators in the evaluation of the performance of the desiccant equipment.

Data should be collected from each desiccant dehumidification system for a period of 90 calendar days during the summer. Additional monitoring in spring, fall, or winter to determine transition season or heating mode performance may also be beneficial. The monitoring should be consistent with the Data Acquisition and Database Management (DADM) standard system monitoring protocol with 15 minute (or less) scan intervals. These intervals should entail, at a minimum, the following measurement points or equivalent points such that system performance, thermal efficiency, and electrical efficiency, can be determined:

1. Outdoor ambient temperature
2. Outdoor ambient relative humidity
3. Building supply air temperature
4. Building supply air relative humidity
5. Heating coil exit temperature
6. Supply air stream pressure drop through system
7. Electrical energy consumed by desiccant unit
8. Regeneration energy consumed by desiccant unit
9. Runtime for each air conditioning unit and desiccant system serving the site
10. Air temperature in the occupied space(s)
11. relative humidity in the occupied space(s).

Note, sensors that need to be placed inside the desiccant unit can be installed by most manufacturers before the unit is shipped. This protects the customer from potentially voiding the warranty due to damage to the equipment that could occur during installation of internal data collection devices. Meters should be included in the design documents for the energy supply lines and installed along with the utility lines.

CASE STUDY

Several desiccant-based systems have been installed at DOD sites. A performance monitoring effort was completed at one site, but no historical utility billing information is available for any of the demonstration sites. This has made even a qualitative analysis of the billing data difficult. Information is available

from one demonstration site, where some of the critical variables were monitored after the desiccant system was installed. The monitoring data include outdoor dry-bulb temperature and relative humidity, process air (supply) dry-bulb temperature and relative humidity, process air flow rate, run time of the unit, regeneration air temperature, electricity consumption, and regeneration gas consumption. The facility, its systems, and the preliminary monitoring data are presented in the following section.

BURGER KING RESTAURANT

The first Army demonstration system was installed at a Burger King restaurant at Aberdeen Proving Ground (APG), MD. Fast food restaurants, large dining facilities and other common areas present a unique situation, because of high occupant density. USACERL wanted to evaluate the use of desiccant-based systems as an air-conditioning solution for such facilities.

The building is an Army-owned Burger King franchise that is representative of a typical fast food restaurant. It is open 24 hours per day, 7 days per week. Several rooftop air-conditioning units serve the building (kitchen, dining area, and bathrooms). The dining area was isolated for this study, because its occupancy density is highest.

Initially, the dining area had two packaged rooftop air conditioning units (5-ton and 7.5-ton) supplying 700 cfm of ventilation out of a total supply flow rate of 5,000 cfm. Although the peak design load matched the equipment nominal capacity (12.5-ton) for the dining area, the components of the load (sensible and latent) did not match the equipment capacities. At the design conditions, the nominal capacity of the two units was reduced from 12.5 tons to 10.5 tons, approximately 13% below the design load (because of supply fan reheat and other losses). The total latent capacity of the units at the design conditions was also less than the required design latent capacity (Meckler et al. 1995). This shortage was exacerbated by off-design conditions in which the latent component of the total load did not drop off nearly as quickly as the sensible component. Because of these problems, the two packaged units were unable to adequately dehumidify and cool the air simultaneously, resulting in frequent hot and humid conditions in the dining area. As a remedy, a nominal 1,600 cfm two-wheel desiccant dehumidification system (TWDDS) manufactured by Engelhard/ICC was installed in the year 1994 as a collaboration between Engelhard/ICC, APG, and USACERL to demonstrate desiccant technology under the Army's Facilities Engineering Applications Program (FEAP).

The installation of the TWDDS was completed in the summer of 1994. Since then, the new system handles the latent load from ventilation and internal gains, and has operated reliably as designed. Improvements in operating conditions were immediately noticed by the restaurant employees and customers. Specifics of the system performance are given below.

EVALUATION OF THE TWO-WHEEL DESICCANT DEMONSTRATION SYSTEM

The objective of this demonstration was to evaluate the cost-effectiveness and energy conservation potential of the TWDDS as it conditioned the air to the appropriate comfort level for the dining area occupants. The design concept was to separate the sensible (internal gains) and latent (ventilation and internal latent) cooling functions. The sensible cooling was handled by the existing 7.5-ton rooftop unit and the latent cooling was accomplished by installing a new TWDDS, which replaced the existing 5-ton rooftop unit. By separating the cooling functions, the effectiveness of the conventional vapor compression system and the desiccant-based system was maximized.

The TWDDS (Figure 3) combines a rotary desiccant wheel with a high-effectiveness rotary heat-exchanger wheel. This combination transfers some of the "sensible penalty" associated with desiccant wheel over to the regeneration air stream. The unit uses a propane-fired boiler for the remainder of the regeneration heat, which is housed within the desiccant unit. The TWDDS operates in a make-up mode (Figure 4). The outside air is passed through the desiccant-wheel where it is dehumidified and then cooled as it passes through the sensible heat wheel. The warm dry air is directed to the conditioned space by its own



FIGURE 3. TWO-WHEEL DESICCANT SYSTEM.

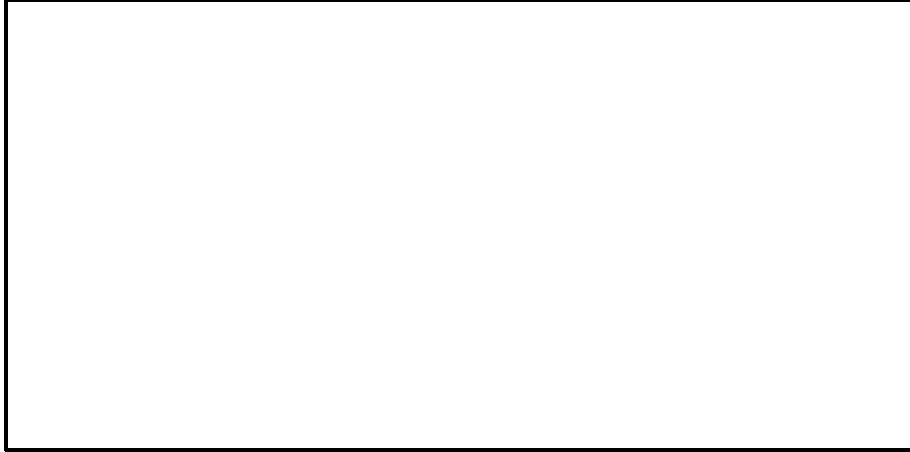


FIGURE 4. DESICCANT SYSTEM OPERATING MODE.

concentric diffuser at ceiling level, and the return air is cooled by the existing 7.5-ton packaged rooftop unit. The dry air from the TWDDS and the cool air streams only mix inside the dining area.

PRELIMINARY MONITORING DATA

Several variables were recorded at 15-minute data intervals from August 1994 through January 1995. Figure 5 shows the daily average outdoor air and process air dry-bulb temperatures for the cooling season. Figure 6 shows the daily average moisture content for outdoor air and process air streams for the cooling season. With the exception of the first 2 weeks of operation, the moisture content of the process air stream stayed between 40 and 60 grains. The daily average electric demand was around 4 kW, and the daily average gas consumption was around 30 cu ft/h.



FIGURE 5. DAILY AVERAGE OUTDOOR AIR AND PROCESS AIR DRY-BULB TEMPERATURES.

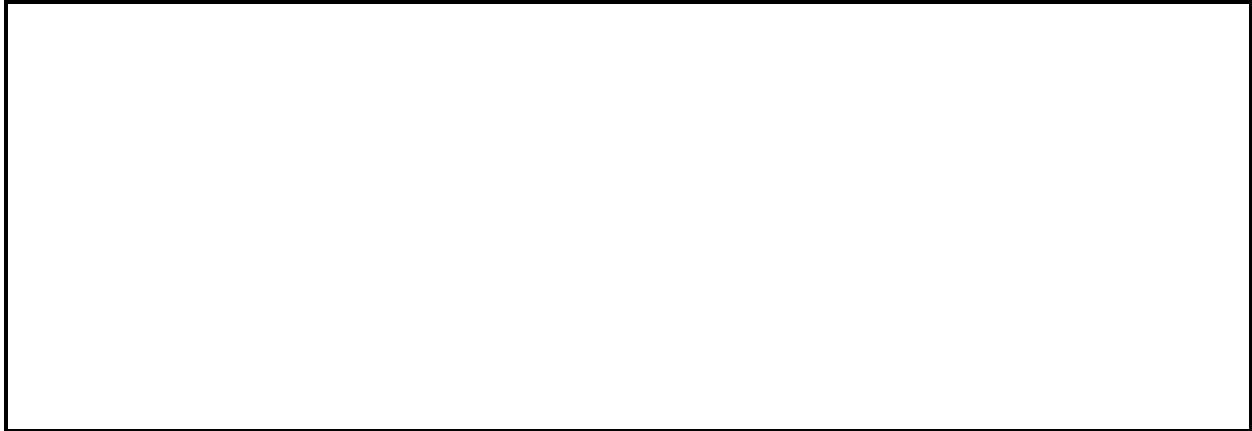


FIGURE 6. DAILY AVERAGE OUTDOOR AIR AND PROCESS AIR HUMIDITY RATIOS.

AUTHOR'S ADDRESS: U.S. Army CERL
P.O. Box 9005
Champaign, IL 61826-9005
CORPSMAIL: CECER-UL-U

REFERENCES AND FURTHER READING

American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. (ASHRAE), "Sorbents and Desiccants," ch 19 in *ASHRAE Handbook: Fundamentals* (Atlanta, GA, 1995).

Burns P R., J.W. Mitchell, W.A. Beckman, "Hybrid Desiccant Cooling Systems in Supermarket Applications," *ASHRAE Transactions*, 91(1B): 457-468 (1985).

Cohen, B.M., and R.B. Slosberg, "Application of Gas Fired Desiccant Cooling Systems," *ASHRAE Transactions*, 94(1):525-536 (1988).

Harriman III, L.G., ed., *The Dehumidification Handbook*, 2d ed. (Munters Cargocaire, Amesbury, MA, 1990).

Harriman, L., *Applications Engineering Manual For Desiccant Systems* (American Gas Cooling Center, Arlington, VA, 1996).

Marciniak T J., R.N. Koopman, D.R. Kosar, "Gas-Fired Desiccant Dehumidification System in a Quick-Service Restaurant," *ASHRAE Transactions*, 97(1):657-666 (1991).

Meckler, M., R. Heimann, J. Fischer, and K. McGaher, *Desiccant Technology Transfer Workshop Manual* (American Gas Cooling Center, Arlington, VA, 1995).